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Comparison of Life Calculations for Oscillating Bearings Considering Individual Pitch Control in Wind Turbines

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Abstract. The fatigue life calculation of bearings under rotating conditions has been well researched and standardized. In contrast, for bearings in oscillating applications no international standards exist. As a result, pitch bearings in wind turbines are designed with different, non standardized approaches. Furthermore, the impact of individual pitch control on pitch bearings has not yet been studied. In this paper four approaches for fatigue life calculation will be applied and compared under individual pitch control conditions. For comparison, the loads and the bearing geometry of the reference turbine IWT 7.5 MW, which is individual pitch controlled, are used. This paper will show how the bearing life calculated by different approaches reacts to individual pitch control conditions. Furthermore, the factors for the modified rating life, according to the ABMA and ISO standards, which implement different operation conditions on the bearings in rotating applications, are calculated for the given loads and the given bearing geometry in oscillating applications.

1. Introduction

A wind turbine with individual pitch control (IPC) moves each blade individually to reduce loads imposed by lift difference. According to theoretical analyses, IPC can reduce the loads on each blade in comparison to blades which are collective pitch controlled [1] [2] [3]. First field tests confirm this theory [4] [5]. The impact on the pitch bearings, which connect hub and blades, has not yet been studied. For first estimations on the impact of IPC the data of a state-of-the-art 7.5 MW reference turbine which is individual pitch controlled are used [6]. The calculated loads of the IWT 7.5 will be presented and used for the life estimation of the pitch bearings. This paper delivers the first theoretical results on how the life time calculations of pitch bearings are influenced under IPC conditions. Furthermore, the comparison of four different approaches will give an overview about the different approaches for fatigue life calculation of bearings under IPC conditions. The *ABMA* and *ISO* standard use different methods to calculate the factors for the modified rating life for rotating applications. In this paper both standards are applied on a pitch bearing under IPC conditions.

2. State of the art - Life calculation

For rotating bearings the calculation of fatigue life has been investigated in depth. The International standard *DIN ISO 281* [7] delivers results which fit well for rolling element bearings. The bearing life, which statistically 90 % of the bearings will survive, can be described by equation (1). In this equation the dynamic load rating C is divided by the equivalent load P. The exponent p depends on the geometry of the rollers.

$$L_{10} = \left(\frac{C}{P}\right)^p \tag{1}$$

The assumptions of this approach are valid for bearings which rotate. For oscillating bearings, new assumptions must be taken into account because under radial load and pure oscillation not all rollers and only subareas of the raceway are loaded. The reversal points of the roller, where the rolling speed decreases, are not considered. Moreover, the physically loaded volume for oscillating bearings is smaller in comparison to rotating bearings. Under this aspect the fatigue life should be greater for oscillating applications. However, the lubrication in oscillating applications worsens, which reduces the fatigue life for oscillating bearings. In the following section, four approaches for the calculation of bearing life in oscillating bearings will be briefly explained. The approaches vary in complexity and in the level of awareness by the industry. All approaches are based on the international standard *DIN ISO 281*. Below, the bearing life under oscillating conditions is named $L_{10,Osc}$.

2.1. DIN ISO 281

The first approach is widespread in the industry and easy to use. The oscillating angle ϕ and the frequency of oscillating motion n_{Osc} are considered in the equivalent speed n by equation (2). The equation is only valid for oscillating angles which are greater than twice of the angular pitch of the rolling elements. For the given example of a pitch bearing in a wind turbine, most oscillating angles will be smaller than the limitation of the equation. Nevertheless, for better comparability between the different models, this model will be applied for all oscillating angles.

$$n = n_{Osc} \cdot \frac{\phi}{180°} \tag{2}$$

2.2. HARRIS 1 [8]

In the first approach published by *HARRIS* [8] a reduced bearing load P_{RE} is used, which depends on the oscillating angle ϕ. This approach is widespread in industry and can be found in several catalogs of bearing manufactures:

$$P_{RE} = \left(\frac{2\phi}{180°}\right)^{1/p} P \tag{3}$$

The purpose of the reduced bearing load P_{RE} is to take the oscillating movement into account. In fact, this approach will lead to the same results as the first presented approach. It is unimportant whether the angle is taken into account in the reduced load P_{RE} or in the equivalent speed n.

$$L_{10,Osc} = \left(\frac{C}{P_{RE}}\right)^p \tag{4}$$

2.3. HARRIS 2 [9]

Furthermore, *HARRIS* developed another approach for oscillating conditions, which corrects the dynamic load rating C_{Osc} depending on the oscillating angle ϕ and the number of rolling elements for each row Z [9]. Therefore, a critical angle ϕ_{crit} is included [10]. If the angle is smaller than the critical one, every roller overruns a subarea that no other roller overruns.

γ is defined as $D_w \cdot \cos(\alpha)/D_{pw}$. The physically stressed volume, which is smaller in comparison to rotating applications, is therefore considered in a simplified manner. Furthermore, *HARRIS* mentioned, that for $\phi < \phi_{crit}/2$ it is possible that fretting corrosion occurs and that it is advisable to rotate the bearing as often as possible for a better lubrication distribution to the rolling elements. This approach is part of the *DESIGN GUIDELINE 03* of *NREL* [9] and is therefore well known in wind turbine engineering. For $\phi > \phi_{crit}$ the results are equal to the results of the *DIN ISO* and *HARRIS 1* approach.

$$C_{Osc}(\phi > \phi_{crit}) = \left(\frac{180°}{2\phi}\right)^{1/3} \cdot C \tag{5}$$

$$C_{Osc}(\phi < \phi_{crit}) = \left(\frac{180°}{2\phi}\right)^{3/10} Z^{0,033} \cdot C \tag{6}$$

$$\phi_{crit} = \frac{360°}{Z(1 \pm \gamma)} \tag{7}$$

2.4. HOUPERT [11]

Another approach is suggested by *HOUPERT*, who calculates the bearing life of an oscillating bearing with a factor A_{Osc}. The factor A_{Osc} represents the ratio of the bearing life during continuous rotation and the bearing life in oscillatory application and is calculated as a function of the load zone parameter ϵ and the oscillating angle ϕ. This approach is the most complex of the four presented. A detailed description of A_{Osc} can be found in *HOUPERT* [11].

$$L_{10,Osc} = A_{Osc} \left(\frac{C}{P}\right)^p \tag{8}$$

HOUPERT advises not to use the model for small oscillating amplitudes. The limitation of this model is set on $\phi < 2\pi/Z$, because *HOUPERT* assumes that smaller values will lead to roller failure from wear [12] instead of rolling contact fatigue [13]. For IPC most values will be smaller than the limitation of the model. Nevertheless, for better comparability between the different models, this model will be applied for all oscillating angles.

3. Proceeding

To compare the four explained approaches under individual pitch control conditions a reference bearing and loads are needed. The approaches will be compared on the individual pitch controlled reference wind turbine IWT 7.5 [6]. Details on the pitch control concept can be found in [5] and [14].

3.1. Pitch bearing IWT 7,5 MW

The pitch bearing of the IWT 7.5 MW was designed by the Fraunhofer IWES in close cooperation with the wind and bearing industry. To create as realistic as possible conditions, a double row four point-contact ball bearing will be used. This bearing is often used for pitch applications [15]. Table 1 shows the main bearing geometry. More detailed data of the bearing will be published in the future. For the presented comparison the given geometric data are sufficient. Figure 1 shows the CAD-Model of the bearing.

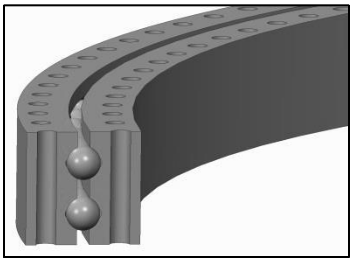

Figure 1: CAD-Model – Pitch bearing

Table 1. Used bearing geometry

Parameter	Size
Pitch diameter $\boldsymbol{D_{pw}}$	4650 mm
Ball diameter $\boldsymbol{D_w}$	80 mm
Contact angle $\boldsymbol{\alpha}$	45°
Number of balls per row $\boldsymbol{Z}$	156
Number of rows $\boldsymbol{i}$	2

For the comparison of the approaches the dynamic load rating of the bearing has to be determined. According to *DIN ISO 281* it is necessary to differentiate between radial and axial bearings by considering the contact angle α of the bearing:

$$Radial\ bearing\colon 0° < \alpha \leq 45°$$
$$Axial\ bearing\colon 45° < \alpha < 90°$$

Thus, the given bearing geometry, with a contact angle of 45° should be calculated as a radial bearing. Depending on the bearing type, the following equations (9) and (10) must be used in accordance with *DIN ISO 281*. The equations consider the bearing material, the geometry, the contact type and empirical factors.

$$C_r = 3{,}647 \cdot b_m \cdot f_{c,r} \cdot (\mathrm{i} \cdot \cos(\alpha))^{0{,}7} \cdot Z^{2/3} \cdot D_w^{1{,}4} \tag{9}$$
$$C_a = 3{,}647 \cdot b_m \cdot f_{c,a} \cdot (\mathrm{i} \cdot \cos(\alpha))^{0{,}7} \cdot \tan(\alpha) \cdot Z^{2/3} \cdot D_w^{1{,}4} \tag{10}$$

With the given equations and the bearing data of table 1 the radial dynamic load rating C_r is 2340 kN and the axial dynamic load rating C_a is 3570 kN. In equation (9) and (10) the bearing geometry-material factor f_c is used. This factor depends on the geometry and also on the bearing type. For the axial geometry-material factor, values from the *DESIGN GUIDELINE 03* [9] are used. The difference to the values from [7] and [16] is that the values of the *DESIGN GUIDELINE 03* are made for large slewing bearings and groove raceway conformity.

In Industry most fatigue life calculations for double row four point-contact ball bearings for pitch applications consider the axial dynamic load rating, independently of the specifications of *DIN ISO 281*. Furthermore, the *DESIGN GUIDELINE 03* of *NREL* advises to use this load rating [9]. The reason for this procedure is that the pitch bearing is highly axially loaded by the wind. Of course, the bearing is also radially loaded from wind and gravity forces acting on the blades, but the radial loads are small in comparison with the axial loads. Anyhow, the contact angle of the bearing α under load is larger than the given value of 45°. The contact angle grows with axial displacement and misalignment which due to the axial and radial loads and the bending moment occur [8] [17]. Therefore, the comparison of the named approaches will also use the axial bearing capacity.

3.2. Loads

Standard pitch bearing lifetime calculations according to *DIN ISO* are done on the base of Load Revolutions Distribtutions (LRD). In their most simple form, LRDs consist of pitch movements that are summed for a number of load classes. Other load signals may be added, as well as pitch speeds. The number of classes in the LRD increases exponentially with every added signal. A LRD will always discard parts of the information and lead to a more conservative calculation. As it is not possible to take into account oscillating movements when using a LRD, a different approach is used in this work. The dynamic loads of the IWT 7.5 reference turbine are simulated with HAWC2. These simulations follow the provisions of [18]. Both fatigue and extreme Design Load Cases (DLC) have been taken into account. The HAWC2 data output is transferred to MATLAB for post processing purposes. The single simulations are combined with the wind speed distributions and the number of special events in the turbine's lifetime to obtain the 20-year loads. The oscillating movements of the pitch bearing are analysed, taking into account range and mean values of the single oscillations as well as the load situation during these movements. The number of cycles and their amplitudes are derived by a range pair count as a rainflow counting algorithm is not apt for this application [19]. Table 2 shows the cycle counting. The calculation of P is given in equation (11) and will be explained in the next chapter. With these cycle counting results, the subsequent calculations are executed.

Table 2: Cycle counting output of the IWT7.5 reference turbine

i	Amplitude range [deg]	No. of cycles	Operation time of active pitch t_i [%]	Mean Amplitude ϕ_i [deg]	Mean Frequency f_i [Hz]	P_i [kN]
1	0,05 - 0,55	2,27E+07	14,88	0,22	0,67	9153,89
2	0,55 - 1,05	4,28E+06	4,59	0,75	0,41	8148,75
3	1,05 - 1,55	2,53E+06	3,75	1,30	0,30	7287,11
4	1,55 - 2,05	2,89E+06	5,18	1,80	0,24	7066,84
5	2,05 - 2,55	3,40E+06	7,51	2,30	0,20	6902,72
6	2,55 - 3,05	3,86E+06	8,96	2,80	0,19	6776,68
7	3,05 - 3,55	4,18E+06	10,04	3,30	0,18	6802,57
8	3,55 - 4,05	4,43E+06	10,89	3,80	0,18	6699,16
9	4,05 - 4,55	4,52E+06	11,39	4,30	0,17	6579,67
10	4,55 - 5,05	3,87E+06	9,98	4,79	0,17	6413,73
11	5,05 - 90	4,80E+06	12,82	5,92	0,16	6380,32

3.3. Analytical Approach

In the following part the application of the different approaches to the load range will be shown. The procedure varies in complexity for the different approaches. First, all time steps without influence on the fatigue life of the bearing are deleted to save computing time. These are all time steps without activity of the pitch controller. In detail 63,83 % of the load steps are deleted. *SHAN* [5] showed in his analytical analyses of pitch bearings, which he compared with field tests, that the loads which occur during the turbine standstill do not influence the fatigue life of the bearing. In a real turbine system there are at any time micro movements which influence the pitch bearing and the bearing life. In an analytical approach it is currently not possible to consider these movements and loads. The radial loads F_r, axial loads F_a and the bending moments M_x and M_y are summed up, as presented in *DESIGN GUIDELINE 03* [9]:

$$P = 0{,}75 \cdot F_r + F_a + \frac{\sqrt{(M_x{}^2 + M_y{}^2)}}{d_{pw}/2} \tag{11}$$

For all approaches the equal load P_{ea} is calculated with equation (12):

$$P_{ea} = \left(\frac{\sum_{i=1}^{i=n} P_i^p \cdot f_i \cdot t_i \cdot \phi_i}{\sum_{i=1}^{i=n} f_i \cdot t_i \cdot \phi_i}\right)^{1/p} \tag{12}$$

The approaches now uses equation (13) with the result of equation (12):

$$L_{10,osc} = \left(\frac{C}{P_{ea}}\right)^p \tag{13}$$

The result of equation (13) is given in oscillations multiplied with 10^6. To gain a result in hours the frequency f and the operation time t need to be considered. The equivalent speed of oscillation n_{Osc} is given by equation (14) and is used for the *HARRIS* approaches. The *DIN ISO* approach furthermore uses equation (2) to calculate the equivalent speed n.

$$\mathrm{n_{Osc}} = \sum_{i=1}^{i=n} f_i \cdot t_i \tag{14}$$

Thus, the bearing life can be expressed in hours:

$$L_{10h,osc} = \left(\frac{C}{P_{ea}}\right)^p \cdot 10^6/(n_{Osc} \text{ or } n \cdot 60) \tag{15}$$

Equation (15) does not consider that the turbine does not pitch at all times. The continuous energy output of 7.5 MW of the turbine is given at wind speeds between 11 m/s and 25 m/s. In this wind speed region the pitch control is active to control the power output. Furthermore, the pitch is active intermittently under typical operating conditions much below the rated wind speed, because due to turbulence there are periods with instantaneous wind speed above rated. Also the wind cyclic individual pitch control for negating the effect of wind shear is taken into account. The pitch is active in 36,17 % of the turbine life. To express the bearing life in wind turbine applications, these effects need to be considered.

$$L_{10,WEA} = \frac{L_{10h,osc}}{0{,}3617} \tag{16}$$

Some turbines have a control region where rated rpm is reached, but power is below rated. In this region the pitch control is active despite the lower power. This concept is not considered in the current controller and therefore not considered in the bearing life calculation. In the future this feature will be taken into account. The bearing life will decrease little with this feature due to the higher number of cycles. With the equations (2), (11), (12), (14) and (15) the bearing life according to *ISO 281* can be calculated. For the *HARRIS 1* approach the reduced load P_{RE} needs to be considered. This can be done similar to equation (12) with the difference that equation (3) is applied. Both *HARRIS* approaches consider for equation (15) the results of equation (14).

$$P_{RE} = \left(\frac{\sum_{i=1}^{i=n}\left(\frac{2\phi_i}{180°}\right)^{1/p} P_i \cdot f_i \cdot t_i \cdot \phi_i}{\sum_{i=1}^{i=n} f_i \cdot t_i \cdot \phi_i}\right)^{1/p} \tag{17}$$

For the approach of *HARRIS 2* the dynamic load rating for oscillating applications C_{Osc} needs to be considered. C_{Osc} can calculated, depending on the oscillating amplitude ϕ_i with the equations (5) and (6).

$$C_{Osc} = \sum_{i=1}^{n=i} C_{osc,i} \cdot t_i \tag{18}$$

The *HOUPERT* approach is complex and in detail difficult to implement in an algorithm. In *HOUPERT's* paper a table is given to easily calculate A_{Osc} for each discrete time step. The algorithm uses the given table and the integral value for the calculated load zone factor ϵ:

$$\epsilon = \frac{1}{2}\left(1 + \frac{\delta_a \cdot \tan(\alpha)}{\delta_r}\right) \tag{19}$$

With the given table of *HOUPERT* and equation (19) the oscillating factor for each time step can be calculated. The factors need to multiplied with the calculated loads for each time step.

4. Results

The results summarized in Table 3 show the extent of the differences of the presented approaches for the bearing life calculation of bearings under IPC conditions. The calculated bearing load P_{ea} for the used time series is 6800 kN. The approach of *ISO 281* thus delivers a bearing life $L_{10,Osc}$ of 5100 h. The bearing life in the considered wind turbine application $L_{10,WEA}$ is 1,6 years. With the approach of *HARRIS 1* a reduced bearing load P_{Re} of 2100 kN is calculated. Thus, the bearing life $L_{10,Osc}$ for this approach is 5100 h. The approaches of *ISO 281* and *HARRIS 1* deliver equal results, because it does not matter if the conversion is effected via the load or the speed. The *HARRIS 2* approach yields a bearing life of 7560 h or a turbine bearing life of 2,4 years. The *HOUPERT* approach delivers a bearing life of 1752 h which is equal to a turbine bearing life of 0,6 years.

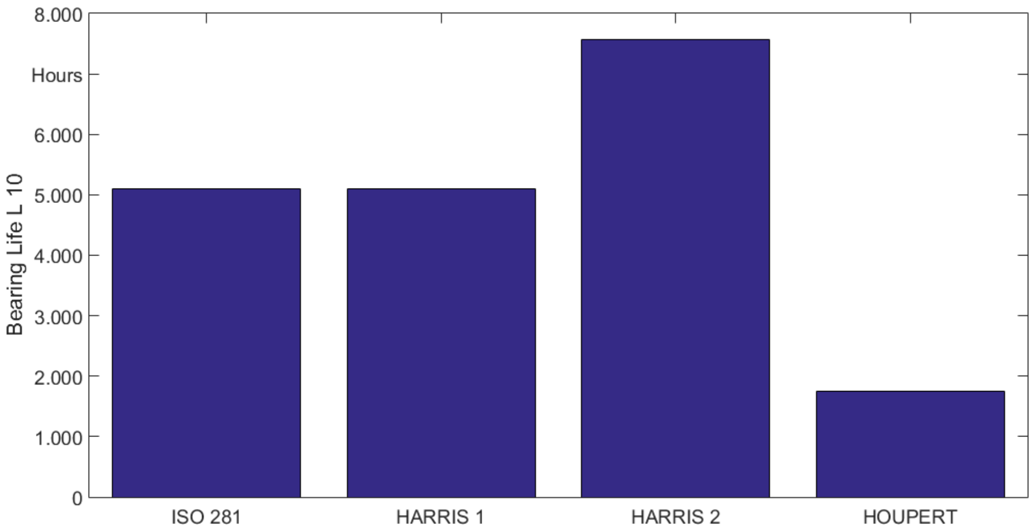

Figure 2: Comparison of the four approaches

Table 3. Result of comparison

Approach	$L_{10,Osc}$	$L_{10,WEA}$ [a]	P_{ea}	P_{Re}	ϕ_{Krit}	$\sum A_{Osc} \cdot t_i$
DIN ISO 281	5098 h	1,6 a	6818 kN	-	-	-
HARRIS 1	5098 h	1,6 a	-	2100 kN	-	-
HARRIS 2	7561 h	2,4 a	6818 kN	-	2,5°	-
HOUPERT	1752 h	0,6 a	6818 kN	-	-	11,8

[a] $L_{10,WEA}$ considers that the pitch control of the wind turbine is active only in 36,17 % of the time.

5. Further effects of bearing life

The results in table 3 consider standard conditions which do not fit with the conditions that occur in wind turbine application. The lubricant in a pitch bearing will not behave under standard conditions and not at any time an EHL-Contact will be present. Furthermore, the lubricant will be influenced by dirt or other negative effects. The structural component's stiffness and the material properties also need to be considered. The *ISO 281* [7] and the *ABMA 9* [16] designate factors which estimate the influence of these effects. It must be mentioned, that the factors are designed for rotating applications. For oscillating applications assumptions must be made, which can be found in the *DESIGN GUIDELINE 03*. In the following chapters the bearings life which consider the effects according to the *ISO* and *ABMA* with assumptions of the *DESIGN GUIDELINE 03* are called $L_{10,Osc,ISO}$ and $L_{10,Osc,ANSI}$.

5.1. Modified L_{10} according to ABMA 9 [16]

According to the American Bearing Manufactures Association (ABMA) the modified bearing life can be calculated with different factors which are multiplied with the bearing fatigue life under standard conditions. For the used pitch bearing these are the factors a_1, a_2 and a_3. Furthermore, *ZARETSKY* [20] applies additional factors. Therefore, the factor a_4 is added for the given pitch bearing.

$$L_{10,Osc,ANSI} = a_1 \cdot a_2 \cdot a_3 \cdot a_4 \cdot L_{10,Osc} \tag{20}$$

The factor a_1 describes the considered reliability. In the case of L_{10} it is equal to 1. For a conservative example of 99 % (L_{99}) it is 0,25. a_2 considers the material. For the reference pitch bearing the steel type 100Cr6 with an hardness of HRC 58 is used. Therefore, the factor a_2 is chosen to be 1. The factor a_3 considers the lubrication. Because of the oscillating movement of the bearing and the small oscillating amplitudes most of the time there are no fully established lubricating films. Thus, a_3 is equal to 0,1 [9]. The factor a_4 considers the stiffness of the adjacent structure. In this case the blades and the hub. *HARRIS* estimates a value of 0,85 for a_4 [9]. The hub of the IWT 7.5 MW has a hub diameter of around 5,5 meters, to ensure that the blades are not affected by the decreased wind in front of the generator. Figure 3 shows a FE-Analysis of the hub. The greatest deformation has a value of more than 9 mm under extreme loads according to *GL* [21]. For the used supporting structure the value of a_4 is chosen to 0,5. With these four factors the modified bearing life can be calculated:

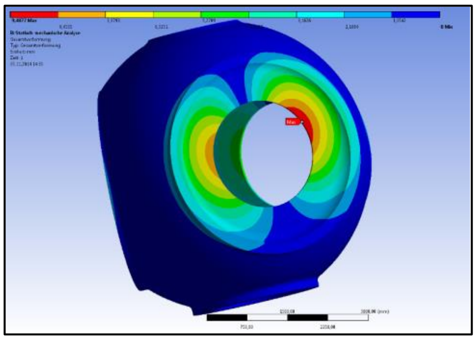

Figure 3: Deformation of hub

$$L_{10,Osc,ABMA} = 1 \cdot 1 \cdot 0{,}1 \cdot 0{,}5 \cdot L_{10,Osc} = 0{,}05 \cdot L_{10,Osc} \tag{21}$$

5.2. Modified L_{10} according to DIN ISO 281 [7]

According to *DIN ISO 281* and assumptions of the *DESIGN GUIDELINE 03* all these effects, which can reduce the bearing life, are implemented in a life factor called a_{ISO}:

$$L_{10,Osc,ISO} = a_1 \cdot a_{ISO} \cdot L_{10,Osc} \tag{22}$$

This factor can calculated with equation (23) and multiplied with the bearing life of each approach to consider the further named aspects. The exponents x_1, x_2, e_1,e_2, e_3 and e_4 are exponents which consider empirical knowledge. κ considers the lubrication conditions. Lubricant contamination is considered through the factor η from equation (24), with the constants c_1 and c_2 to calculate the contamination factor. To consider the material properties and the given loads the fatigue load limit P_u, which can calculated with *DIN ISO 76* [22] or *ABMA 9* [16], and the further calculated dynamic equivalent load P, are also given in the equation. As mentioned before, assumptions need to be considered to calculate a_{ISO} for oscillating applications, cause the *ISO* do not considers oscillating movements.

$$a_{ISO} = 0{,}1\left[1 - \left(x_1 - \frac{x_2}{\kappa^{e_1}}\right)^{e_2} \left(\frac{\eta P_u}{P}\right)^{e_3}\right]^{e_4} \tag{23}$$

$$\eta = 0{,}173 \cdot c_1 \kappa^{\,0{,}68} {d_m}^{\,0{,}55} \left(1 - \frac{c_2}{{d_m}^{\,1/3}}\right) \tag{24}$$

For the given bearing and the given conditions a_{ISO} can be calculated with the mentioned assumptions to ~0,1:

$$L_{10,Osc,ISO} = 0{,}1 \cdot L_{10,Osc} \tag{25}$$

6. Conclusions

The results show that none of the approaches lead to results which accomplish the required turbine life of 20 years. Furthermore, it becomes clear that the different approaches show variations in the results. Under some operating conditions the oscillating amplitudes of the bearing are smaller than the limitations of the presented approaches, which influence the results. Therefore, the results are only valid for the comparison. For the amplitudes which are smaller than the limitations, new approaches need to be investigated. Furthermore, the occurrence of false brinelling / fretting corrosion needs to be further explored to predict the bearing life in the field.

The given cycle load shows the conditions the pitch bearing needs to withstand. The equivalent load P_{ea} of 6800 kN is very high in comparison to the dynamic capacity of the bearing C_a of 3500 kN. The effects of a modified, improved bearing on the results will be a part of future research. Furthermore the turbine pitch is active 16 times each minute. This high pitch rate leads to around 1000 pitch cycles each hour.

The *ISO 281* approach lead to equal results as the *HARRIS 1* approach. The *ISO 281* pursued the idea to convert the oscillation into revolution via the speed. The *HARRIS 1* approach converts via the equivalent load. The results show, that it does not matter which transformation is used, because both yield equal results.

The other approach of *HARRIS* follows the same idea as the first presented approaches. In the *HARRIS 2* approach the axial dynamic capacity is calculated depending on the oscillating amplitude ϕ. The new calculated capacity $C_{a,osc}$ of 13250 kN leads to the highest calculated bearing life. The approach considers a simplified stressed volume of the raceway. Furthermore, the approach is easy to use for collective loads, because the transformation is done via the capacity of the bearing.

The last compared approach of *HOUPERT* use a factor A_{osc}. This factor establishes a relationship between rotating and oscillating bearing life. 34 % of the oscillating amplitudes are smaller than the limitation of the model. The result of *HOUPERT* is therefore only valid for the comparison.

There is little information about the conditions of pitch bearings in the field. Especially IPC has not been thoroughly investigated. The estimations for the bearing life factors according to the american and even the international standard showed, that the calculation currently is very conservative. The american standard leads to an estimated bearing life which is just 5 % of the calculated life under standard conditions. The international standard leads to a value of 10 % of the calculated life under standard conditions. Most of the influences which decrease these life factors are estimated or based on empirical data which do not fit for oscillatory applications. To get more accurate results, more research in life factors for wind turbine applications is needed.

At least, further damage mechanisms like wear, which also occur in pitch bearings are not be considered. The approaches of *HARRIS* and *HOUPERT* advise to use big oscillating amplitudes, to avoid false brinelling / fretting corrosion. For the given conditions 15 % of the oscillating amplitudes are smaller than 0,55 °, so that wear damages are likely to occur.

This paper shows that tests of pitch bearings are needed to prove different bearing life approaches, as calculatory measures do not provide certain results and predicable operational experience with IPC controller is not yet available. The financial implications of necessary bearing changes can not be estimated at this point, but will be subject to future research.

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